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ENHANCEMENT OF CONVECTIVE HEAT TRANSFER IN CHANNELS BY MEANS OF ARTIFICIAL FLOW TURBULENCE

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ENHANCEMENT OF CONVECTIVE HEAT TRANSFER IN CHANNELS BY MEANS OF ARTIFICIAL FLOW TURBULENCE

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ABSTRACT

Evaluation of the enhancement of convective heat transfer by turbulent flows through profile, roughened, and corrugated (finned) channels. The evaluation is based on an analysis of the structure of a two-layer turbulent flow. An equation containing three empirical constants is derived to characterize the limiting turbulence of a flow composed of a layer with a constant coefficient of turbulent heat exchange and a viscous sublayer in which the molecular viscosity is greater than its turbulent viscosity. Artificial roughness is found to increase heat removal in channels by a factor of 1.5. The reduction of heat exchanger dimensions by diminishing the exchanger diameter, by increasing heat carrier velocity, or through a wider use of finned surface is suggested.

Two-layer model of the structure of turbulent flow, characterizing /123* its limiting turbulence is discussed. On the basis of the model analysis,

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* Note: Numbers in the margin indicate pagination in the original foreign text.

conclusions are reached concerning the perspectives of the enhancement of heat transfer.

The problem of the enhancement of convective heat transfer, which is related to the reduction in metal enclosures and in the over-all size of numerous heat exchangers used in power engineering, chemistry, and other segments of the national economy, is of great interest. One of the methods of such enhancement is the production of flow turbulence. For this purpose, devices are used which provide periodic expansions and disruption of the flow, and as a result periodic restoration of the boundary layer (profile channels, artificial roughness, various coverings, etc.). These devices also produce intense oscillations and pulsations in the flow with a varying frequency spectrum (Ref. 1).

The thermal resistance as a function of the Prandtl number is very nonuniformly distributed along the normal to the flow. If one uses the usual three-layer model of the turbulent flow structure in a channel, as has been done for instance in (Ref. 2) (a viscous sublayer, intermediate region, turbulence nucleus), then the calculations done for $Re = 10^4$ show that at $Pr = 0.72$ the thermal resistance of the viscous sublayer constitutes 32.3% of the total thermal resistance, that of the intermediate region - 52%, and that of the turbulence nucleus - 15.7%. Here the thermal resistance is given by the formula

$$R = r_s + r_i + r_n = 1/Nu$$

where Nu is the Nusselt number, r_s, r_i, r_n denote, respectively, the thermal resistance of the sublayer, of the intermediate region, and of the nucleus of the flow. It is very interesting to note that the main part of the air thermal resistance is concentrated, not in the viscous sublayer, but in the intermediate region, a notable fraction of the total thermal resistance being concentrated also in the nucleus of the turbulent flow. For Pr = 10 one obtains, respectively, r_s = 74.5%, r_i = 22%, r_n = 3.3%. For Pr = 200 we have r_s = 99%.

Therefore, in order to achieve enhancement of convective heat transfer, having in mind complete turbulent motion, for gas flow it is, strictly speaking, necessary to render the entire region of the boundary layer turbulent, whereas for the flow of viscous liquids it is enough to set in turbulent motion only the viscous sublayer. It must be assumed that it is practically impossible to render a flow without any limit turbulent, and that its limiting turbulent state exists.

Recent measurements in boundary layers have shown (Ref. 3) that the /124 turbulent flow near the walls can, even for the pre-disruption state, be divided into the viscous sublayer (0.001 - 0.01 δ), the region of the logarithmic velocity profile (0.1 - 0.2 δ), and a region with a constant coefficient of turbulent exchange (0.9 - 0.8 δ), where δ is the thickness of the boundary layer.

According to the experiments with strongly roughened tubes (Ref. 4), the turbulent viscosity changes only slightly along the normal to the surface.

The analysis of the experimental data shows that the turbulence of the flow leads to an expansion of the region with a constant coefficient of turbulent exchange.

The limiting turbulent state can be represented as the region with a constant coefficient of turbulent heat exchange expanded up to the viscous sublayer (the region of the logarithmic profile is "washed out"). Starting with the above preliminaries, we shall analyze a two-layer model of flow in a channel involving a region with a constant coefficient of turbulent viscosity and a region of a viscous sublayer. The viscous sublayer (the region in which the molecular viscosity is greater than its turbulent viscosity) will exist at any practical turbulence in the nucleus of the flow; however, the thickness and the level of turbulence in it will vary. According to (Ref. 2), the turbulent viscosity in the sublayer is given by the formula

$$\mu_T = \beta v_* \rho \frac{y^4}{y_1^3}, \tag{1}$$

where μ_T is the coefficient of turbulent viscosity; $\beta = 0.032$ is an experimentally determined coefficient; v_x denotes dynamical velocity; ρ is the density; y denotes the distance from the wall; y_i is the thickness of the viscous sublayer.

Defining, as is customary, the thickness of the sublayer by the expression $y_i = \alpha\nu/v_*$, we shall obtain the following for the velocity at the boundary of the sublayer

$$u_1 = \int_0^{y_1} \frac{\tau dy}{(\mu + \mu_T)} = \int_0^{y_1} \frac{\tau_W}{\mu(1+\beta\alpha)} dy = \frac{\alpha v_*}{1+\beta\alpha}, \tag{2}$$

where α is a proportionality coefficient; τ denotes the tangential stress; τ_W denotes the tangential stress at the wall; μ is the coefficient of the dynamical viscosity.

Dimensional analysis considerations, on which the derivation of the structure formulas for the viscous sublayer is based, are also valid for a turbulent flow. An increased turbulence is taken into account by the magnitude of the dynamical velocity $v_* = \sqrt{\tau_w/\rho}$.

Let us assume that the numerical values of the constants α and β are not changed. We shall consider stabilized flow in a straight, smooth, circular tube of radius R_0. The tangential stress over the cross section of the tube is, as we know, expressed by the formula $\tau = \tau_w (R/R_0)$, where τ_w is the friction stress at the wall, R- is the radius (at which the stress is being calculated) of the tube. We shall assume that the coefficient of turbulent viscosity μ_T is constant over the cross section of the tube (except for the region of the sublayer) and along its length. Then the equation for determining the velocity assumes the form

$$\mu_T \frac{du}{dR} = -\tau_w \frac{R}{R_0} \tag{3}$$

or $u = C - \frac{\tau_{0T}}{\mu_T R_0} \frac{R^2}{2}$. We determine the constant C from the condition that the velocities be equal at the sublayer boundary. According to equation (2) we have

$$C = \frac{a v_*}{1+\beta a} + \frac{\tau_w}{2\mu_T R_0}\left(R_0 - a\frac{\nu}{v_*}\right)^2,$$
$$u = \frac{a v_*}{1+\beta a} + \frac{\tau_w}{2\mu_T R_0}\left(R_0 - a\frac{\nu}{v_*}\right)^2 - \frac{\tau_w}{\mu_T R_0}\frac{R^2}{2}. \tag{4}$$

We consider that the magnitude of μ_T is determined, just as for free stream jets, by the formula (Ref. 5): /125

$$\mu_T = \sigma\sqrt{\rho J_0} = \frac{\sigma\sqrt{\pi}}{2}\,\mathrm{Re}, \tag{5}$$

where σ is an empirical constant which can be determined by experiments with stream jets, J_0 is the intensity of the stream jet.

Using the relations $\frac{v_*}{\bar{u}} = \frac{\sqrt{\lambda}}{2\sqrt{2}}$ -- where λ is the resistance coefficient, and $\bar{u}$ denotes the mean velocity -- and (Ref. 5), we rewrite equation (4) in the form

$$u = \bar{u}\left[\frac{\sqrt{\lambda}\,\alpha}{2\sqrt{2}(1+\beta\alpha)} + \frac{\lambda}{16\sigma\sqrt{\pi}}\left(1 - \frac{4\sqrt{2}\,\alpha}{\sqrt{\lambda}\,\mathrm{Re}}\right)^2 - \frac{\lambda}{16\sigma\sqrt{\pi}}\left(\frac{R}{R_0}\right)^2\right]. \tag{6}$$

Three empirical constants are included in the above expression for the limiting turbulent velocity profile; one of them, σ, is determined from experiments with free stream jets, and the constants α and β -- from experiments with smooth tubes under the usual conditions of turbulent flow. The quantity σ characterizes the initial turbulence to a certain extent. The value $\sigma = 0.21$ was used for further calculations (Ref. 5). The constant α characterizes the dimensionless thickness of the viscous sublayer $\alpha = y_1 v_*/\nu$. As follows from an examination of the experimental data, in boundary layers of various kinds (Ref. 3), including diffusive boundary layers, and also in sections, in pre-disruption states, the value $\alpha = 6$ can be considered as very close to its minimal value.

The velocity profile determined by expression (6) can be represented by the power law

$$\frac{u}{u_{max}} = \left(\frac{y}{R_0}\right)^{1/n}. \tag{7}$$

The Re number is included in the expression for the velocity (6). However, calculations show that its influence on the velocity profile is negligible. Thus, when Re is changed from 10^4 to infinity, $u/\bar{u}$ changes by less than 0.1%.

Therefore, the value of Re is barely apparent in the exponent n in formula (7). The value $n \cong 50$ obtained for the case under consideration is valid for all Re numbers.

The turbulent velocity profile, analyzed above, fills to a greater extent than usual velocity profiles for turbulent flow in a tube. Thus, according to the data in (Ref. 6) for $Re = 3 \cdot 10^6$, n = 10. The increase in the value of n is explained by a greater flow turbulence.

Let us determine the mean velocity

$$\bar{u} = \frac{1}{\pi R_0^2} \int_0^{R_0} 2\pi R u dR = \frac{\alpha v_*}{1+\beta\alpha} + \frac{\tau_W}{2\mu_\tau R_0}\left(R_0 - \alpha \frac{\nu}{v_*}\right)^2 - \frac{\tau_W}{4\mu_\tau} R_0. \tag{8}$$

After simple transformations, using equation $v_*/\bar{u} = \sqrt{\lambda}/2\sqrt{2}$, we obtain from equation (8) the equation which allows us to determine the coefficient of hydraulic resistance for turbulent flow through a tube

$$\frac{\lambda\alpha^2}{(1+\beta\alpha)^2} = 2\sqrt{2} - \frac{\lambda}{4\sqrt{2}\pi\sigma}\left[\left(1 - \frac{\alpha 4\sqrt{2}}{\sqrt{\lambda}\,\mathrm{Re}}\right)^2 - \frac{1}{2}\right]. \tag{9}$$

Taking $\sigma = 0.21$, $\alpha = 6$, we obtain $\lambda = 0.3$. Similarly to $u/\bar{u}$, λ barely depends on the Reynolds number. The conclusion stated above is very remarkable, since it is usually believed that the self-similarity of the stream flowing close to a wall is determined by the presence of pressure resistance -- as is the case, for instance, with a flow in greatly roughened tubes. However, the result obtained above -- namely, that in the absence of pressure resistance (with only friction resistance being present) the resistance is independent of the Reynolds number -- has also been confirmed by recent experiments with tubes whose interior was roughened /126 by a series of rings placed along these tubes (Ref. 7). The pressure distribution was measured on ring-like transverse partitions placed in the tubes by means of a drain-pipe arrangement. The pressure on the front surface of a partition was, naturally, greater than that on its reverse side. However, a value for the pitch of the ring-like partitions was found in the experiments at which the swelling created by the subsequent partition neutralized the decrease in pressure, and the form resistance was found to be zero. The resistance coefficient is also in this case independent of the Reynolds number. Thus, in the presence of high turbulence self-similar, undisrupted flows may exist in the vicinity of the walls. We may determine the heat transfer in a tube for stabilized conditions using our two-layer model of turbulent flow, with the heat load being constant over the length of the tube. For this purpose we use the Lyon integral (Ref. 2).

$$\frac{1}{\mathrm{Nu}} = 2\int_0^1 \frac{\left(\int_0^\xi \omega\xi\,d\xi\right)^2}{\left(1+\varepsilon\,\mathrm{Pr}_\tau\frac{\mu_\tau}{\mu}\right)\xi}\,d\xi, \tag{10}$$

where $\xi = \frac{R}{R_0}$, $\omega = \frac{u}{\bar{u}}$, $\varepsilon = \frac{\lambda_\tau}{rc\mu_\tau}$, λ_τ is the coefficient of turbulent heat conduction, c is the heat capacity.

According to the adopted two-layer model, we have the following two integrals:

$$\frac{1}{2\mathrm{Nu}} = \int_0^{\xi_1} \frac{\left(\int_0^\xi \omega\xi\,d\xi\right)^2}{1+\varepsilon\,\mathrm{Pr}\,\sigma\,\mathrm{Re}\frac{\sqrt{\pi}}{2}}\,d\xi + \int_{\xi_1}^1 \frac{\left(\int_0^\xi \omega\xi\,d\xi\right)^2}{1+\varepsilon\,\mathrm{Pr}\,\beta\frac{v_* R_0}{\nu}\frac{(1-\xi)^4}{(1-\xi_1)^3}}\,d\xi. \tag{11}$$

where

$$\xi_1 = 1 - \frac{\alpha}{\mathrm{Re}}\sqrt{\frac{32}{\lambda}}.$$

The turbulent viscosity in the sublayer is taken into account by equation (1). Determining ω from equation (6), we obtain the following after integrating to within an accuracy of 0.1% (for Re $\geq 10^4$)

$$\frac{1}{2\mathrm{Nu}} = \frac{0{,}0625}{1 + \varepsilon \Pr \sigma \mathrm{Re}\frac{\sqrt{\pi}}{2}}\left(M^2 - \frac{M\lambda}{24\sigma\sqrt{\pi}}\right)\left(1 - \frac{\alpha 4\sqrt{2}}{\mathrm{Re}\sqrt{\lambda}}\right)^4 +$$
$$+ \frac{\alpha}{\mathrm{Re}}\left(\frac{M^2}{4} - \frac{M\lambda}{64\sigma\sqrt{\pi}}\right)\sqrt{\frac{32}{\lambda}}\left(\ln\frac{a^2 + a\alpha\sqrt{2} + a^2}{a^2 - a\alpha\sqrt{2} + a^2} + \right. \qquad (12)$$
$$\left. + 2\,\mathrm{arc\,tg}\frac{a\alpha\sqrt{2}}{a^2 - \alpha^2}\right)\frac{1}{\sqrt{1024\beta\alpha \Pr}},$$

where

$$M = \frac{\sqrt{\lambda}\,\alpha}{2\sqrt{2}(1 + \beta\alpha)} + \frac{\lambda}{16\sigma\sqrt{\pi}}, \quad a = \sqrt[4]{\frac{\alpha^3}{\beta \Pr}}.$$

Figure 1 shows the dependence Nu = f(Re) for Pr = 0.72; this was calculated from formula (12), where we took λ = 0.30; α = 6, β = 0.032; σ = = 0.21; ε = 1. The results of the calculation based on formula (12) for the case ε = 2 in the nucleus of the flow, as is usually assumed in a theory of stream jets, are represented by a dotted line. /127

As we see from the graphs, the effect of the change in ε within the indicated limits on the heat transfer is negligible, this being related to the small thermal resistance of the flow nucleus in the case under consideration. The limiting value of the heat transfer obtained for Re = 10^4 is 13.0 times greater, and for Re = $5 \cdot 10^4$ is 15.5 greater than that obtained for the usual turbulent flow through a smooth tube.

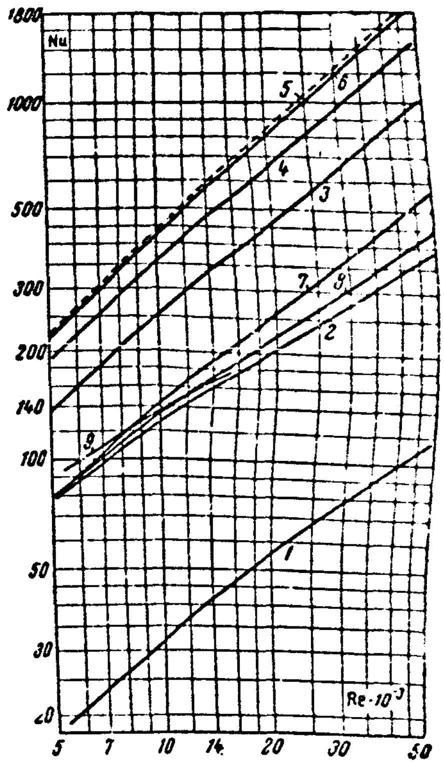

Figure 1

Heat Transfer in Turbulent Flows: 1- $Nu = 0.02\ Re^{0.8}$; 2- dense layer (Ref. 11), Pr = 0.72 (recalculated for u (narrow cross section); 3- from formula (14), Pr = 20; 4- from formula (14), Pr = 100; 5- from formula (12), $\varepsilon = 2$; 6- from formula (12), $\varepsilon = 1$; 7- from formula (14), Pr = 0.72; 8- roughened tube (Ref. 9), $k/R_0 = 0.4$; L/K = 9.8; 9- heat transfer at the critical point (Ref. 12).

The heat transfer, determined by formula (12), is obviously difficult to achieve in practice. For this purpose, it is necessary to render the flow turbulent continuously along the entire length of a channel, without at the same time decreasing the friction stress in the vicinity of the wall. The turbulent flows encountered in practice are different, mainly in the fact that the flow is made turbulent due to periodical expansions and disruptions of the flow, which diminishes the friction stress near the wall. The flow in dense granular layers, in greatly roughened tubes, in various coverings, etc., may serve as examples of such flows.

In the periodically disrupted flows under consideration, the friction stress near the wall in diffusive sections is diminished, and at the points of disruption it strives to zero. The energy dissipation in this case takes place mainly at a certain distance from the wall. In this connection, we shall examine the results of experiments carried out by Fedge and Faulkner (Ref. 8), who have measured the distribution of the friction stress around the entire perimeter of the cylinder. The mean friction stress calculated from fromula

$$c_f = \left(\int_0^S \tau_w \, dS \right) \Big/ \left(\frac{1}{2} \rho \bar{u}^2 \pi d \right)$$

for Re = $\bar{u}\pi d/\nu = 5.2 \cdot 10^5$, where S -- the perimeter line, d -- the diameter of the cylinder, was found to be $c_f = 0.00318$. For a smooth plate and the same Re number, we have for a turbulent boundary layer c = 0.005, and for a laminar boundary layer c = 0.002. Thus, in spite of an intense disruption process, $*\tau_w$ proves to be similar to the case of a flow with vanishing gradient past a smooth surface. A direct determination of τ_w, with a sufficient accuracy, for the case of a dense layer, greatly roughened tube, etc., is at present impossible, whatever the theoretical methods used, and its experimental determination is extremely difficult. By analogy with the examined case of interrupted flow through a cylinder, we can assume in the first approximation that in a dense layer and in greatly roughened tubes the friction stress in the vicinity of a wall is the same as that for a smooth surface.

In the examples under consideration, for instance, in a dense layer, new boundary layers start forming behind the points of flow separation from the surface; movement in the boundary layers in the rear regions is directed towards the main flow and is regulated by the return flow. Thus, in contrast to a developed movement through a tube, in our case the movement will be periodically unstable. /128

Similar phenomena take place also in greatly roughened channels. Instead of such periodically unstable motion, one can, to a certain approximation, examine movement which is stable on-the-average in a tube. Based on the above statements, we assume that the average friction stresses at the wall $\bar{\tau}_w = \frac{1}{S}\int_0^S \tau_w dS$, where S is the line of the channel perimeter -- with periodical disruptions being present -- will differ negligibly from the stresses encountered in flow through a smooth channel.

Therefore, in order to give an estimate for limiting, turbulent heat transfer with periodic disruptions of the flow, we assume in formula (12) that for a viscous sublayer τ_w is the same as that for the case of a smooth tube, that at the boundary of the sublayer τ undergoes a jump, and that in the nucleus of the flow $\lambda = 0.30$ is the value corresponding to τ. In spite of the fact that the thickness of the viscous sublayer is assumed to be the same as that for a smooth channel, we assume that the turbulence in the sublayer is increased, and that, consequently, the vertical component of the pulsating velocity at the boundary of the sublayer is

$v_y' \sim v_* = \sqrt{\frac{\lambda}{8}}\,\bar{u}$, where λ = 0.30. Then, at the sublayer boundary the turbulent viscosity is given by the formula

$$\frac{\mu_T}{\mu} = \beta a \sqrt{\frac{\lambda}{\lambda_0}}. \qquad (13)$$

The subscript 0 corresponds to ordinary turbulent flow.

Taking into account the stated premises, on the basis of formula (12), we obtain the expression

$$\frac{1}{2\mathrm{Nu}} = \frac{0{,}0634\left(1 - \frac{248}{\mathrm{Re}}\right)}{1 + \mathrm{Pr}\,\mathrm{Re}\cdot 0{,}186} + \\ + \frac{2{,}23}{\mathrm{Re}\sqrt{\lambda_0}\sqrt[4]{\mathrm{Pr}}\sqrt{\frac{\lambda}{\lambda_0}}}\left(\ln\frac{36 + 8{,}5a + a^2}{36 - 8{,}5a + a^2} + 2\,\mathrm{arctg}\,\frac{8{,}5a}{a^2 - 36}\right), \qquad (14)$$

where

$$a = \frac{9{,}07}{\sqrt{\mathrm{Pr}}\sqrt[8]{\frac{\lambda}{\lambda_0}}}.$$

Figure 1 presents the dependence Nu = f(Re), calculated from formula (14) for Pr = 0.72; 20; 100. We shall compare the obtained results with the available experimental data on turbulent flows (dense layer, greatly roughened tubes). At large values of both the relative roughness k/R_0 and the distances between the elements of ring-like roughness, flow through such a tube can be schematically represented as an expansion of a stream jet in a limited space. For this kind of arrangement, at Pr = 0.72 the maximal heat transfer was observed at k/R_0 = 0.4; L/k = 9.8 (Ref. 9), where k is the height of protuberances, L is the distance between them.

These data are represented in Figure 1.

Now we shall examine the experimental data on a dense layer. The /129 boundary layer which is turbulent to a maximal extent at separate cross sections of the layer, due to the action of vortex tracks, and the phenomena in a dense layer or in the roughened tube discussed above, are very similar to each other. In both cases, the nucleus of the flow (external streamline flow) is saturated with vortices. As we know, the data on heat transfer in a dense layer, due to the uncertain determination of particle location, usually pertain to the so-called diffusion velocity -- namely, to the velocity which would exist if no particles were present. In order to determine the real velocity in dense layer channels, it is necessary to find the real distribution of particles in the layer. Sleduya (Ref. 10) assumes that, from the statistical point of view, a dense layer composed of mixed identical spheres may be regarded as a certain combination of separate groups of densely-packed, simple cubical distributions correlated in the ratio resulting in the observed *.

For a simple cubical packing, the porosity P = 0.476. Correspondingly, for densely-packed distributions (pyramidal and tetrahedral) P = 0.26. On the basis of experimental data, the mean porosity of the layer can be assumed to be P = 0.4.

In accordance with the adopted assumptions, we have an equation determining the ratio of the corresponding packings $0.4 = 0.476x + 0.26(1-x)$. Hence, $x = 0.648$. For a simple cubical packing, the ratio of the narrow

* Translator's note: Word illegible in foreign text, but probably porosity.

transitional cross section to the total cross section is $F_{nr}/F = 0.212$, whereas for densely-packed coverings $F_{nr}/F = 0.0885$. The mean value for the narrow cross-section of the layer, in accordance with the packing fractions, is $F/F_{nr} = 5.93$. The obtained ratio F/F_{nr} allows us to determine the velocity in the narrow cross section of the layer from the diffusion velocity; by the tube bundle analogy, the former is the controlling velocity. We may also recalculate the data on the heat transfer. Figure 1 in the above-mentioned paper presents the data obtained by W. H. Denton (Ref. 11).

The comparison between the computational results based on formula (14) and the experimental data on a dense layer and greatly roughened tubes, shown in Figure 1, makes it clear that these data are very similar to each other. Such a coincidence allows us to assume that both the theoretical dependence and the experiment characterize a state which is very close to the limiting state, from the point of view of turbulence in the presence of periodic expansions and disruptions of the flow in the vicinity of the wall.

The intensity of heat transfer at $Re = 10^4$ is in the case under consideration 4.5 times greater than that for ordinary turbulent flow through a tube. Generally speaking, the maximal heat transfer in a flow will take place in the case of the minimal thickness and maximal turbulence of the boundary layer. Obviously, flow at the critical point of, for example, a cylinder under the condition of maximal turbulence of the advancing flow corresponds to the stated conditions.

Figure 1 shows the data for the examined case, obtained in (Ref. 12) for initial turbulence which was the greatest achieved in the experiments (the distance between the cylinder and the grating which produces turbulence is $L/d = 6$, where d is the diameter of the cylinder). As can be seen from the graph, these data for the explored range of Re numbers are very close to the data on the dense layer in a greatly roughened tube, although the structure of the boundary layer is different in this case (laminar boundary layer), and the friction stress at the wall is not diminished.

The case of a turbulent boundary layer at the front critical point does not occur in practice. The agreement with the calculations according to formula (14) is in this case accidental; it shows, however, that the computation based on formula (12) is close to the maximal heat transfer observed in practice.

A comparison of the computational results based on formulas (12) and (14), and also a comparison with the data for ordinary turbulent flow through a tube, indicates that periodic expansions and disruptions, and, consequently, restorations of the boundary layer, substantially intensify heat transfer. However, the enhancement of heat transfer would have been even /130
greater [formula (12)] if these phenomena had not been followed by a decrease in the friction stress at the wall, as compared with τ in the stream. Nevertheless, this phenomenon is an essential property of disruptive flows. Creation of equivalent turbulence in a flow not due to expansions and local disruptions at the wall, but due to some other mechanism -- without simultaneously decreasing the friction stress at the

wall, will lead to a further enhancement of heat transfer. Up to the present, this problem has not been solved in practice. The hydraulic resistance, determined by equation (9), does not contain a fraction corresponding to pressure resistance, as is the case for a dense layer and for roughened tubes, and, consequently, it is minimal for a flow with the heat transfer close to its limiting value.

In this connection, it is of interest to compare the obtained data [heat transfer from equation (14), resistance from equation (9)] with the available experimental results for various devices producing turbulence, and also with ordinary turbulent motion through a tube, the comparison being made with regard to energy. In Figure 2, such a comparison is shown in the coordinates α, N, where N is the power needed to overcome the resistance, taken with respect to the unit area. The figure also shows the results of experiments with optimal spiral turbulence-producing devices (Ref. 13), and with roughened profile channels (Ref. 14). It follows from the graphs that the limiting turbulence due to restoration of the boundary layer and local disruptions ensures the 2.3 increase in heat removal over the value found for ordinary turbulent flow through a smooth tube, with resistance losses being equal.

The data on artificial roughness shown here provide for the 1.5 increases in heat removal, as compared with the value for a smooth tube. Thus the "reserve" increase in heat removal, taking into account the practical unattainability of the limiting state (from the point of view of the relation between the resistance losses and the heat removal), is found to be quite small.

(a)

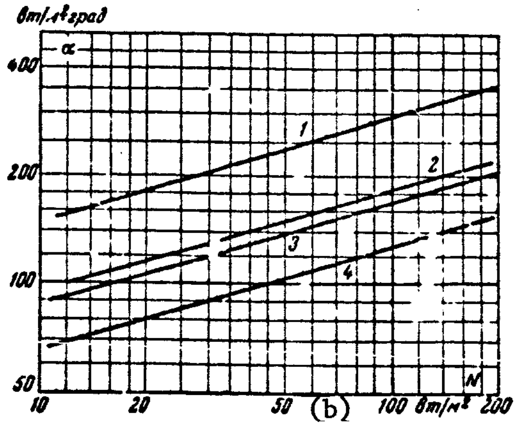

Figure 2

Comparison Between Turbulent Flows Taking into Account the Energy Relations: 1- from formulas (9) and (14); 2- spiral turbulence-producing devices (Ref. 13); 3- profile channel (Ref. 14); 4- $Nu = 0.02\ Re^{0.8}$, $\lambda_0 = 0.316/\sqrt[4]{Re}$.

(a) - w/1^2·degree; (b) - 100 w/m^2.

Considerable effort has recently been directed toward creating turbulent surfaces of heat transfer due to the optimal form of the channel walls.

The above analysis shows that the possibilities in this direction are very unpromising from the point of view of heat removal with equal resistance losses.

Considerable reserves for enhancement of heat transfer in forced /131 turbulent motion, in addition to those mentioned earlier, can be found, apparently, in the application of dispersive materials existing in forced states (pseudo-liquified layer, etc.).

The increase in compactness of heat exchangers should also be achieved by a wider use of finned surfaces, by diminishing the channel diameters, and by increasing heat carrier velocity.

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